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TO: Dr. D. Mills
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INFRASONIC ENHANCEMENT OF HEAT TRANSFER - PRELIMINARY REPORT

B.G. Woods

ABSTRACT

Claims have been made that infrasound can be used to significantly reduce the electricity consumption in a convection heating process (up to 94% reduction/3/) by enhancing the rate of heat transfer. In response to these claims, preliminary lab tests were performed to evaluate the electricity savings potential of infrasound. The results from the tests indicated an increase in the convective heat transfer with infrasound, but the laboratory-scale device used did not yield electricity savings. The poor electrical performance is a result of the low electricity-to-infrasound conversion efficiency in the lab unit. A more efficient infrasound transducer may be able to reduce the electricity consumption in convection heating processes.

INTRODUCTION

Pressure oscillations below the audible sound frequencies (eg, below 20 Hz) are known as infrasound. The velocity oscillations which are induced by infrasound can be used to enhance convective heat transfer. Infrasound is one of a number of heat transfer enhancement techniques, which have the potential to reduce the electrical energy consumed in a convective heat transfer process. Since convective heat transfer is widely used in industry, there could be a considerable amount of electricity conservation potential for an effective enhancement technique.

The general field of heat transfer enhancement can be divided into two categories; 1) passive devices, in which no external power is required, and 2) active devices, in which external power is required. Within these categories, a number of enhancement techniques have been developed, as listed in Table 1. Of the techniques listed in Table 1, fluid vibration is of particular interest for the following reasons:

1) Fluid vibration does not require changes to the heat transfer surface and thus can be applied to many different convection processes (as opposed to many of the other techniques which have certain requirements of the heat transfer surface, or product being processed).

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2) The published literature indicates that oscillating flow (or fluid vibrations) can provide a significant enhancement of convective heat transfer [2].

3) A Swedish manufacturer has commercialized the enhancement technique in the infrasonic range of frequencies. This will help to provide an easier and less expensive transfer of the technology.

Table 1 - Techniques for Enhancing Forced Convection Heat Transfer (from ref. 1)

Passive (no external power required)	Active (external power required)
- roughened surfaces	- mechanical aids (for stirring fluids)
- extended surfaces	- surface vibration
- displaced flow devices	- fluid vibration (eg. via sound)
- swirl flow devices	- electrostatic fields
- coiled tubes	- jet impingement
- additives for liquids	- injection or suction
- additives for gases	

The mechanism by which sound (and infrasound) enhances heat transfer is not well established. However it is generally accepted that the sound changes the fluid flow regimes in the boundary layers of the heat transfer surface. A summary of the hypothesized mechanisms for heat transfer enhancement is presented in reference [2].

In the literature published on the influence of sound on heat transfer, there is a lack of comparative information on the energy consumption during forced convection with and without sound. Therefore, lab tests were performed to provide a preliminary assessment of the potential for using infrasound as a technique for reducing the electricity consumption in forced convection processes.

RESULTS FROM THE LAB TESTS

The results from the lab tests indicate that when infrasound is added to steady air flow, the convection heat transfer is significantly increased. In certain conditions, the rate of heat transfer was double the rate in steady air flow. However, as the air flow rate was increased, the degree of heat transfer enhancement decreased. These results are summarized in the following paragraphs, and further details on the lab apparatus and the test results are presented in the Appendices.

In order to evaluate the potential for reducing electricity consumption with infrasound, a power substitution ratio (SR_{el}) was defined, which represents the amount of fan power that can be replaced by a unit of infrasound power. This is represented in equation form as

$$SR_{el} = \Delta P_F / P_{IS,e}$$

where ΔP_F is the reduction in electrical power used to provide steady air flow
$P_{IS,e}$ is the electrical power consumed by the infrasound equipment to maintain the same heat transfer, with the fan power reduced.

When the substitution ratio (SR) is greater than 1.0, the reduction in fan power is greater than the additional power used for the infrasound, and therefore there is a net reduction in electricity consumption. Conversely, when the substitution ratio is less than 1.0, there is a net increase in electricity consumption.

Since the efficiency of the laboratory-scale equipment was anticipated to be less than the efficiency of a production infrasound unit, a second ratio (SR_{ac}) was defined which uses the acoustic energy in the airstream. This ratio indicates the upper limit on the electricity savings that could be realized by a more efficient infrasound unit. The ratio SR_{ac} is defined as

$$SR_{ac} = \Delta P_F / P_{IS,a}$$

where $P_{IS,a}$ is the acoustic power in the airstream near the heat transfer surface.

Listed in Table 2 are some typical values for the power substitution ratios obtained in the lab tests. From Table 2, the ratio SR_{el} was much less than 1.0, therefore the reduction in fan electricity consumption is much less than the additional electricity used by the lab infrasound unit. This indicates a net increase in electricity consumption. However, the ratio SR_{ac} was much greater than 1.0, which means that a more efficient infrasound unit could provide a reduction in electricity consumption.

Table 2 - Typical Values for the Power Substitution Ratios from the Lab Tests

Convection Coefficient ($W/m^2{}^oC$)	Power Substitution Ratios [1]	
	SR_{el}	SR_{ac}
100	0.03	86
111	0.05	106

1 calculation of the substitution ratios shown in appendix B.

A quick test was also performed on a passive device for inducing pulsed air flow. The device was essentially a stirrer driven by the steady air flow. By introducing the passive device to the air flow, the heat transfer rate increased by 15%.

CONCLUSIONS

The lab tests discussed in this report are preliminary in nature, however the following conclusions can be made.

1) Under certain steady flow conditions, infrasound can increase the rate of heat transfer by a factor of two.

2) The efficiency of the infrasound unit tested was very low (approximately 0.04%), therefore it could not be used to reduce the total electricity consumption in a forced convection process. However, a more efficient method for generating the infrasound, if available, could achieve a net reduction in the electricity consumption.

RECOMMENDATIONS

The tests indicate that infrasound is an effective technique for enhancing convective heat transfer. However, the low efficiency of the laboratory unit prohibited the demonstration of the electrical load reduction concept. Therefore, in order to demonstrate the load reduction concept, and also broaden the scope of heat transfer enhancement tools for the Energy Management Branch, the following extensions to this work are recommended.

1) Determine if more efficient methods of converting electricity into infrasound are available, and estimate the electric power saving potential.

2) Investigate the further design and use of passive devices for inducing pulsations in an airstream, and estimate the electric power saving potential.

3) Review other heat transfer enhancement techniques that can be used in process heating, and estimate their relative importance with respect to electric power savings.

Approved by:

A.F. Baljet
Manager
Electrical Research Department

Submitted by:

Brian Woods

B.G. Woods
Utilization Engineer
Utilization Section

Reviewed by:

J.T. Strack
Unit Head - Process Applications
Utilization Section

BGW:dmc

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APPENDIX A

Configurations of the Lab Test Apparatus

In order to evaluate the potential for using oscillating air flow to enhance heat transfer, a lab scale infrasound unit was purchased. The infrasonic frequency range was used, as claims have been made that infrasound can be used to reduce electricity consumption in forced convection processes by 94% [3]. A sketch of the unit is shown in Figure A1.

At the outlet, or chimney, of the infrasound unit, a cartridge heater was mounted to allow measurements of the convection heat transfer from the heater to the airstream. Knowing the electrical energy dissipated by the heater, the supply temperature of the airstream, the surface temperature of the heater, and the surface area of the heater, the convection coefficient (h) can be calculated as

$$h = Q / [A (T_s - T_a)] \quad W/m^2\,°C \qquad - (1)$$

where Q = steady state rate of energy dissipation in the heater (watts)
A = surface area of the heater (m^2) = 3.6×10^{-3} m^2
T_s = surface temperature of the heater (°C)
T_a = temperature of the air supply (°C)

The energy dissipated in the heater was determined as the product of the current through, and the voltage across the cartridge heater. The surface temperatures were measured with thermocouples attached at the furthest upstream and downstream locations on the cartridge heater. The air temperature was measured in the supply duct, and thus the thermocouples were sheltered from the heater.

Additionally, the air flow rate was measured in the air supply duct using orifice meters, which were calorimetrically calibrated.

In order to use the infrasound in the most efficacious manner, a number of configurations were used to combine the infrasound and steady air flow. These configurations are illustrated in Figures A2 to A4.

The configuration recommended by the equipment manufacturer is the through-flow configuration (Figure A2). In this configuration, the steady and oscillating flows were in line with one another. The main problem with this configuration was that some of the sound energy escaped through the air supply duct, and thus was not available at the heat transfer surface. Screens were used in the exit from the supply duct to minimize this problem, however there was no method available to eliminate the unwanted flow of sound energy in the air supply duct.

Another configuration, which focused all the infrasonic energy on the heat transfer surface, is the cross-flow (closed) configuration shown in Figure A3. The steady flow is transverse to the oscillations in this configuration. The weakness in this configuration is that the oscillatory flow across the back of the cylinder (the side furthest from the infrasound unit) may be inhibited. Therefore the heat transfer enhancement from the infrasound may only affect a portion of the cylinder.

The third configuration tested was the cross-flow (open) configuration shown in Figure A4. The objective in this configuration was to increase the exposure of the cylinder (heater) to oscillatory

flow. The weakness in this configuration is that since the heat transfer surface is in "open" air, the infrasound energy will dissipate to some degree before reaching the heat transfer surface.

After testing the configurations described above, it was apparent that the low efficiency of converting electricity to infrasound was an important factor in the feasibility of using infrasound to enhance heat transfer. Therefore a device which used the steady flow of air to induce oscillations was fabricated and tested. This passive device was intended to provide flow (or pressure) oscillations at the heat transfer surface and is illustrated in Figure A5.

Figure A1 - Infrasonic Test Unit

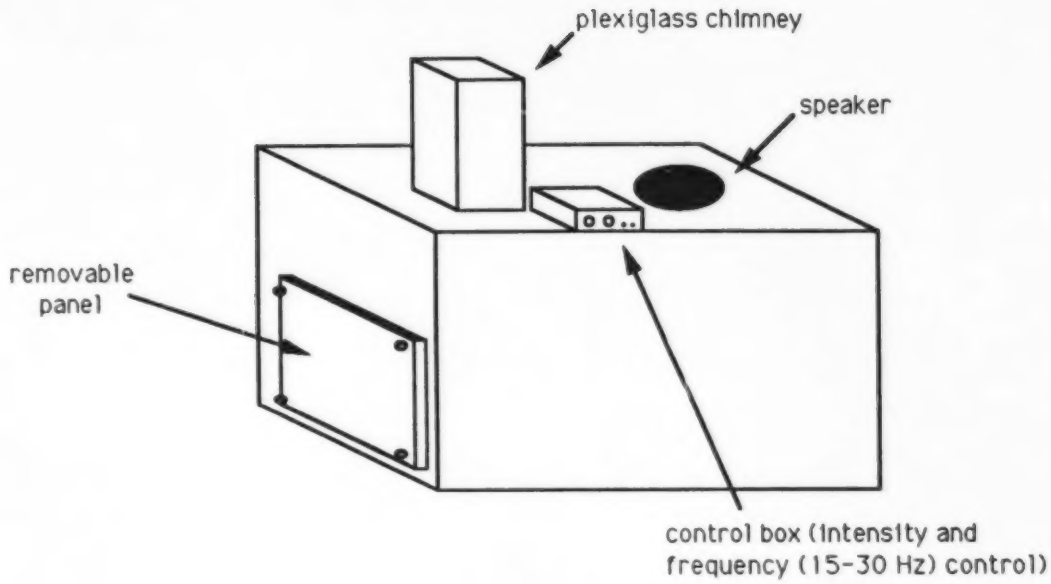


Figure A2 - Configuration for Through-Flow Tests

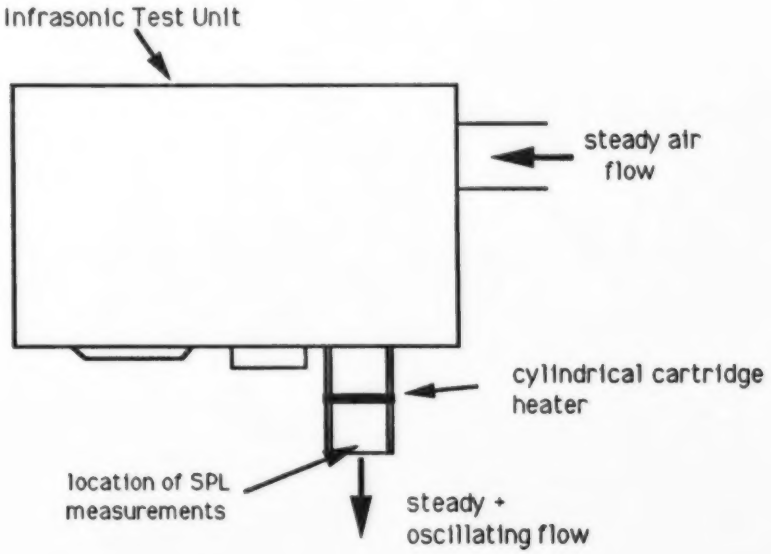


Figure A3 - Configuration for Cross Flow (closed) Tests

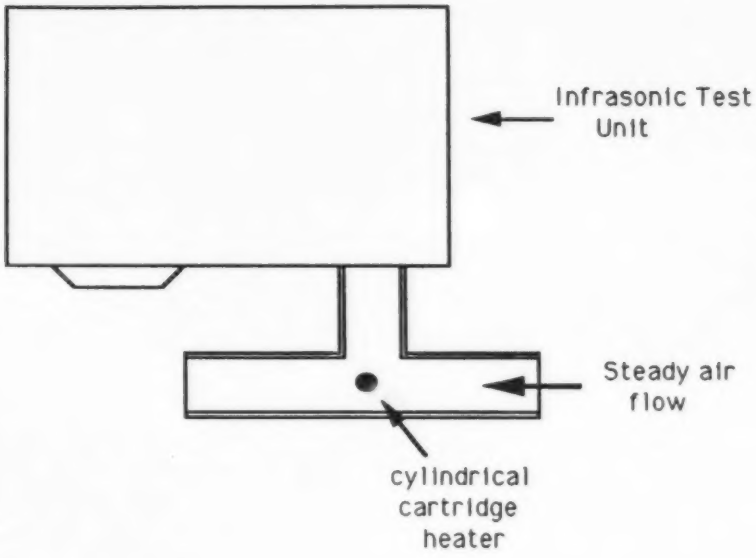


Figure A4 - Configuration for Cross Flow (open) Tests

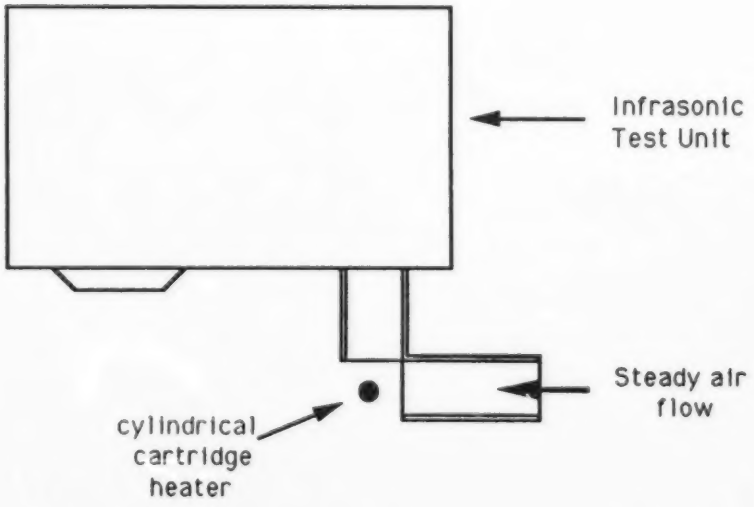
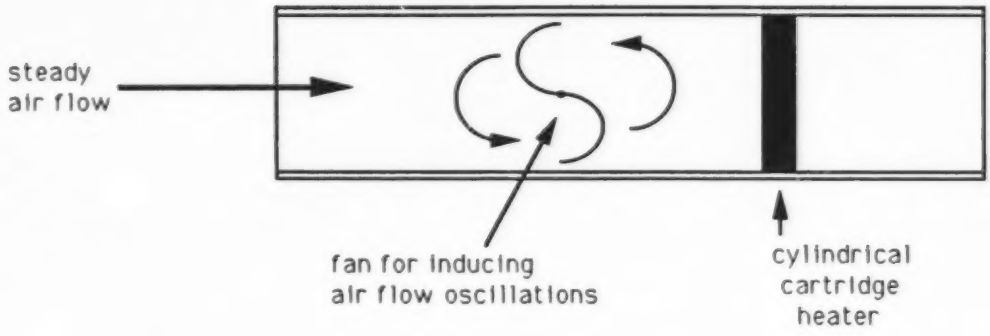


Figure A5 - Passive Device for Inducing Air Flow Oscillations



APPENDIX B

Lab Test Results

In this appendix, the lab test results are presented separately for each apparatus configuration. The largest enhancement of heat transfer was achieved with the "through-flow" apparatus, and therefore there were more tests performed with this configuration, and correspondingly there are more results to present. The test results with the other apparatus configurations are presented briefly to illustrate their relative degrees of heat transfer enhancement.

B.1 Results from Tests with Through-Flow Configuration

The convection heat transfer achieved with and without sound in the through-flow configuration is shown in Figure B1. From Figure B1, it can be seen that the largest enhancement occurred at the lowest air flow rates. This result can also be seen in Figure B2, in which the heat transfer enhancement ratio is plotted. The heat transfer enhancement ratio (R_{HTE}) is defined as

$$R_{HTE} = h_{IS} / h_O \qquad - (2)$$

where h_{IS} = the convection coefficient obtained with infrasound plus steady flow, and
h_O = the convection coefficient obtained with only steady flow.

The effect of the infrasonic frequency on the enhancement ratio is shown in Figure B3. It has been suggested [4] that the enhancement of heat transfer will be the greatest when the fluid oscillating frequency is the same as the vortex shedding frequency. The vortex shedding frequency is dependent on the fluid velocity, and calculated values are shown on Figure B3 for the different steady flow rates used. The vortex shedding frequency (f_{vs}) was calculated using the following equation (from reference [5])

$$f_{vs} = S\, u_\infty / D \qquad - (3)$$

where S is the dimensionless Strouhal number ($\approx$0.2 for the flow rates used)
u_∞ is the velocity of the air flow (m/s)
D is the diameter of the cylinder (m)

It should be noted that the above formula (3) is for a cylinder in steady flow, and the infrasound could have had an influence on the actual vortex shedding frequency.

From Figure B3, it appears that the frequency does have an affect on the enhancement of heat transfer. However, it does not appear to have been affected by the vortex shedding frequencies, which were higher than the infrasonic frequencies used. Also, the different degrees of enhancement can not be explained with the relative sound pressure levels at each frequency (listed in Figure B2). Because of the preliminary nature of the tests, it would be premature to speculate on the actual nature of the frequency effect.

A set of tests was performed to determine the affect of the infrasound intensity on the convection coefficient. This information is needed to estimate the effect of substituting infrasound energy for steady flow fan energy. The results of these tests are shown in Figure B4. The 18 Hz infrasound frequency was used because it yielded the highest degree of heat transfer enhancement. The test result with no applied infrasound was plotted in Figure B4 at 78 dB, which was the measured background noise level with the air flowing. From Figure B4, it appears that the heat transfer rate is very sensitive to the sound pressure level (SPL) within the band of SPL's tested. Also, the slope

of the curve indicates that the degree of heat transfer enhancement would still be significant if the SPL's are increased past the levels used in the lab tests.

The lab results presented in this report represent a preliminary stage in this project, and the emphasis has been to obtain "ball park" figures to allow a preliminary assessment of the potential for infrasound enhancement of heat transfer. In order to estimate the probable error in the results, a comparison with theoretical heat transfer coefficients for steady air flow is presented in Figure B5. The theoretical values for the heat transfer coefficient (h) were obtained by using the measured air and surface temperatures (for the fluid properties), and the air flow rate (and thus velocity) in the following equation (from reference [6] page 216)

$$h = (k_f C / D) (u_\infty D / \nu_f)^n Pr^{1/3} \qquad - (4)$$

where k_f = thermal conductivity of the fluid (W/m°C)
C = a constant determined by the Reynolds number
= 0.683 for flow rates used in lab tests
D = diameter of the cylindrical cartridge heater (m)
u_∞ = fluid velocity approaching the cylinder (m/s)
ν_f = kinematic viscosity of fluid (m^2/s)
n = a constant determined by the Reynolds number
= 0.466 for flow rates used in lab tests
Pr = Prandtl number of the fluid.

As seen in Figure B5, the experimental values are approximately 25 - 50% higher than the theoretical values. Based on the published data, this deviation is likely due to the fact that the heat transfer coefficients were calculated using the average of two temperature measurements around the circumference of the cylinder. It is expected that increasing the number of temperature measurement locations would eventually bring the experimental results close to the theoretical values. Since this magnitude of error would not change the general conclusions from the lab tests, further accuracy in the measurements was not pursued at this stage of testing.

B.1.1 Energy Substitution Effectiveness of Infrasound

Ontario Hydro's interest in infrasound, is in the use of infrasound energy to displace a larger amount of fan energy while maintaining the same amount of heat transfer in a convection heating process. This section attempts to estimate the electrical power trade-off when substituting infrasonic energy for fan energy in a forced convection process.

Since the heat transfer enhancement was the highest in the through-flow configuration, the power substitution effectiveness is evaluated only for this configuration.

The energy input required for the steady flow is estimated by first determining the kinetic power (specific kinetic energy X flow rate) and dividing by a typical fan efficiency of 80%. This approach was taken, as opposed to simply measuring the electricity consumption, because the fan speed in the lab apparatus had to be slowed down to achieve the largest rates of heat transfer enhancement with the infrasound. Since the fan speed was controlled with the voltage, the fan was operating away from the design conditions and thus the electricity consumption would not be representative of typical fans. Therefore, the kinetic power (P_f) in the air stream was determined as

$$P_f = \dot{m} u_\infty^2 / 2 \qquad - (5)$$

but $u_\infty = \dot{m} / \rho A$ - (6)

therefore $P_f = \dot{m}^3 / [2 (\rho A)^2]$ - (7)

where $\dot{m}$ = air mass flow rate (kg/s)
ρ = density of the air (kg/m^3)
A = cross sectional area of the test section (infrasound unit's chimney) (m^2)

Allowing for a typical fan efficiency of 80% (from reference [7], page 14-55), the power required for the steady air flow is

$$P_s = \dot{m}^3 / [1.6 (\rho A)^2] \quad - (8)$$

Since the infrasound unit had an extremely low efficiency, it was desirable to compare the power required for steady flow to the acoustic power in the airstream as well as the electricity consumed by the infrasound. To allow this comparison, the acoustic power was approximated using the sound pressure level (SPL) measurement. In the lab tests, the SPLs were measured in the chimney (see Figure A2), and thus were measured on a plane progressive wave. From reference [8] (page 14), the acoustic power in a plane progressive wave is related to the SPL according to the relation.

$$SPL = 10 \log [W \rho_o c / (A p_{ref}^2)] \quad - (9)$$

$$SPL = 10 \log [W / A] + 10 \log [\rho_o c / p_{ref}^2] \quad - (10)$$

$$10 \log [W / A] = SPL - 10 \log [412.5 / (2 \times 10^{-5})^2] \quad - (11)$$

$$\log [W / A] = (SPL - 120) / 10 \quad - (12)$$

$$W = A \, 10^{(SPL - 120)/10} \quad (watts/m^2) \quad - (13)$$

where SPL = sound pressure level measured in the test section (dB)
A = cross sectional area of the test section (m^2) = 0.0145 m^2
$\rho_o c$ = characteristic impedance of air at 20°C and 760 mm Hg = 412.5 mks rayls
p_{ref} = reference sound pressure = 2×10^{-5} Pa

Therefore, for the apparatus used, the acoustic power in the airstream can be estimated with

$$W = (0.0145) \, 10^{(SPL - 120)/10} \quad (watts) \quad - (14)$$

Using the results from the experiments, the following examples illustrate the electricity substitution potential for infrasound. The analysis in each example works toward a substitution ratio (SR_{el}), which is defined in this report as

$$SR_{el} = \Delta P_F / P_{IS} \quad - (15)$$

where SR_{el} = power substitution ratio for a constant rate of heat transfer
ΔP_F = the reduction in fan electricity consumption
P_{IS} = the electricity consumed by the infrasound unit while maintaining the equivalent rate of heat transfer

The breakeven point for energy substitution occurs when SR = 1. If SR is less than one, the infrasound enhancement will use more electricity than just the steady flow. If SR is greater than one, then infrasonic enhancement will be an effective use of electricity.

A second substitution ratio (SR_{ac}) is also determined which uses the acoustic power in the air stream. The ratio SR_{ac} is used as a way to isolate the effect of the lab unit's low efficiency (see Table B1), on the overall results and conclusions. The ratio SR_{ac} is thus an indication of the upper limit of the electricity savings that could be realized by a more efficient infrasound unit. The ratio SR_{ac} is defined as

$$SR_{ac} = \Delta P_F / P_{IS,a} \qquad -(16)$$

where $P_{IS,a}$ is the acoustic power in the airstream near the heat transfer surface, which maintains the same rate of heat transfer.

Table B1 - Output Power and Electricity Consumption of the Infrasound Unit at 18 Hz

Sound Pressure Level (dB)	Electricity Consumption (W)	Acoustic Power in Air (W)	Conversion Efficiency
114	18	3.6×10^{-3}	0.02%
118	24	9.1×10^{-3}	0.04%
120	32	1.4×10^{-2}	0.04%

Example 1

A heat transfer coefficient of $h = 100$ W/m^2 °C can be achieved, in the through-flow configuration, in either of the following ways:

i) Steady air flow of 69.5 g/s (from Figure B5).

In this test, the surface temperature was 103°C, the average air temperature was 64°C and the density $\rho = 1.04$ kg/m^3 (from $\rho = P/RT = 1.01 \times 10^5/(287(64+273))$). Therefore, using equation (8) (with area $A = 0.0145$ m^2), the power required for the steady flow is

$$P_{a1} = m^3 / [\, 1.6 (\rho A)^2 \,] = (0.0695)^3 / [\, 1.6 (1.04 \times 0.0145)^2 \,] = 0.92 \text{ watts}$$

ii) Steady air flow of 27.5 g/s plus 118 dB infrasound at 18 Hz (from Figure B4).

For a surface temperature of 103°C, the average air temperature was 64°C and thus the air density was 1.04 kg/m^3. The power required for the steady flow of 27.5 g/s is

$$P_{a2} = (0.0275)^3 / [\, 1.6 (1.04 \times 0.0145)^2 \,] = 0.06 \text{ watts}$$

Using equation (14), the acoustic power in the air is

$$W_2 = (0.0145)\ 10^{(118-120)/10} = 0.01 \text{ watt}$$

Considering only the acoustic power, the substitution ratio is

$$SR_{ac} = (P_{s1} - P_{s2})/W_2$$
$$= (0.92-0.06)/0.01$$
$$= 86$$

However, considering the electricity consumption of the infrasound unit (from Table B1), the substitution ratio is

$$SR_{el} = (0.92-0.06)/24 = 0.03.$$

Since $SR_{el} << 1$, the laboratory infrasound unit would not provide a reduction in electricity consumption. However, the potential electricity savings that could be achieved with a more efficient infrasound unit are significant since SR_{ac} is much greater than 1.

Example 2

A heat transfer coefficient of 111 W/m^2 °C may be achieved in either of the following two ways.

i) Steady air flow of 85 g/s (extrapolating from Figure B5 using equation (4), such that $h_1/h_2 = (u_{\infty,1}/u_{\infty,2})^{0.466} = (m_1/m_2)^{0.466}$).

The surface temperature for this heat transfer rate was 95°C, therefore the average air temperature was 60°C and the air density is 1.06 kg/m^3. Using equation (8) the power required for the steady flow of 85 g/s is

$$P_{s1} = (0.085)^3 / [\ 1.6\ (1.06 \text{ X } 0.0145)^2\ ] = 1.6 \text{ watts}$$

ii) Steady air flow of 27.5 g/s plus infrasound at 120 dB and 18 Hz (from Figure B4).

The power required for the steady flow is

$$P_{s2} = (0.0275)^3 / [\ 1.6\ (1.06 \text{ X } 0.0145)^2\ ] = 0.06 \text{ watts}$$

Using equation (14), the acoustic power in the air is

$$W_2 = (0.0145)\ 10^{(120-120)/10} = 0.014 \text{ watts}$$

Considering only the acoustic power in the air, the substitution ratio is

$$SR_{ac} = (\ 1.6 - 0.06\ ) / 0.014 = 106$$

Considering the electricity consumption of the infrasound, (32 watts - from Table B1), the substitution ratio is

$$SR_{el} = (\ 1.6 - 0.06\ ) / 32 = 0.05$$

As in the first example, the laboratory infrasound unit would not yield electricity savings as an enhancement technique since $SR_{el} << 1$. However, the potential savings for a more efficient unit are significant since $SR_{ac} >> 1$.

B.2 Results From the Tests with the Cross-Flow Configurations

As discussed in Appendix A, a problem with the through flow configuration was that a noticeable amount of acoustic energy was escaping down the air supply duct. Two alternatives to the through flow configuration are shown in Figures A3 and A4. The advantages and disadvantages of these two alternatives are qualitatively discussed in Appendix A.

A series of tests at three steady flow rates, and various infrasonic frequencies were performed with the cross-flow (closed) configuration. The results of these tests are shown in Figure B6. By comparing the results of Figure B6 with the results in Figure B2, it can be seen that the enhancement ratio (h_{IS}/h_O) was consistently lower at each frequency with the cross-flow (closed) configuration. A possible explanation for this result is that the opening of the infrasound chimney is perpendicular to the duct which may be inhibiting the oscillating flow across the back of the heater.

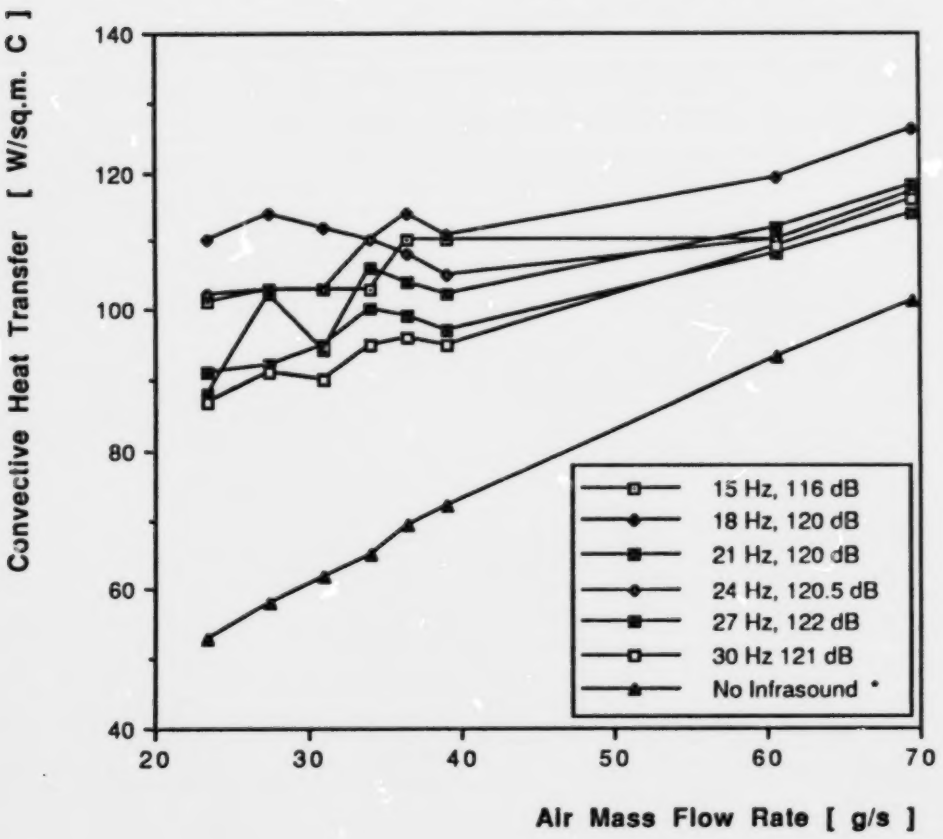
An alternative to the cross-flow (closed) configurations is the cross-flow (open) configuration (shown in Figure A4). The results from a set of tests with the cross-flow (open) configuration are shown in Figure B7. The results in Figure B7 indicate that the cross-flow (open) configuration did not provide a significant heat transfer improvement over the cross-flow (closed) configuration, and thus it is not as effective as the through-flow configuration.

B.3 Passive Device For Inducing Flow Oscillations

The lab results with the infrasonic test unit indicate that the flow oscillations do provide a significant enhancement of heat transfer. However, the process suffers from a low conversion efficiency from electricity into acoustic energy. Although the focus of this project is the enhancement of heat transfer with infrasound, a quick test was performed with a passive device which induced a pulsation of the air flow. The passive device is described in Appendix A.

In the lab tests, the convection coefficient in an airflow of 40 g/s without the passive device, was 67 W/m^2 °C. When the passive device was included in the air flow, the convection coefficient increased to 76 W/m^2 °C, without a noticeable difference in the air flow rate. This represents a 15% increase in heat transfer.

Figure B1 - Convection Heat Transfer with and without Infrasound



* the background noise level was approximately 78 dB without infrasound

Figure B2 - Infrasonic Enhancement of Convection Heat Transfer

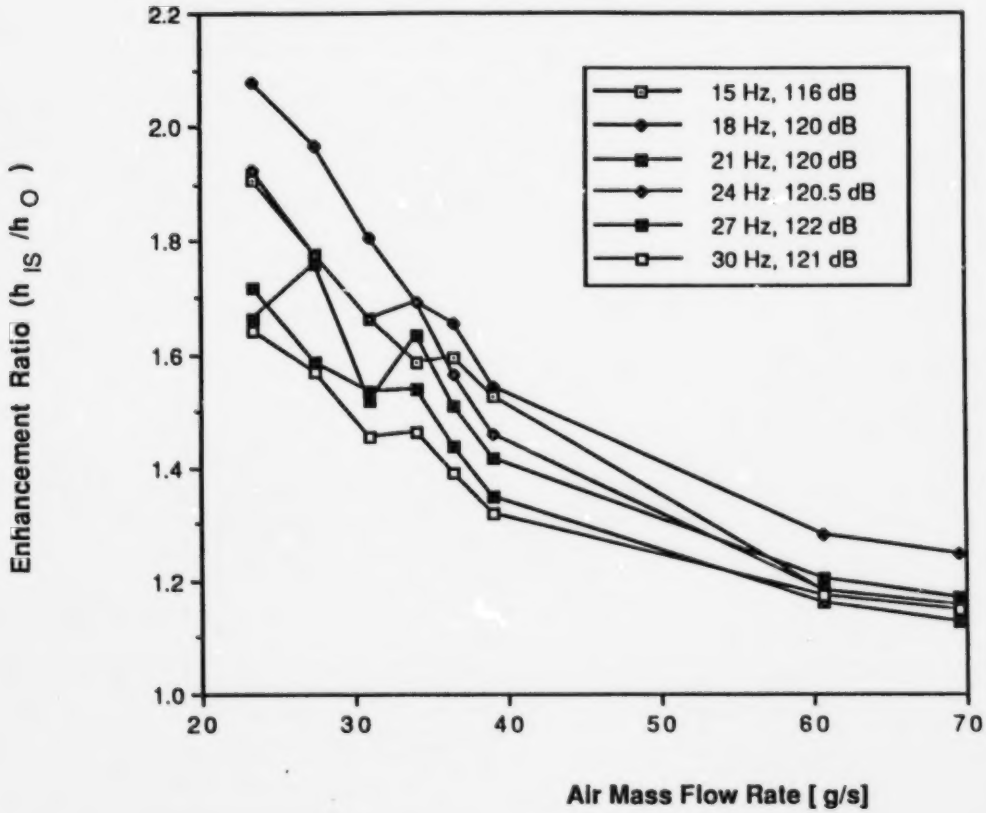


Figure B3 - Influence of Infrasound Frequency on Heat Transfer Enhancement

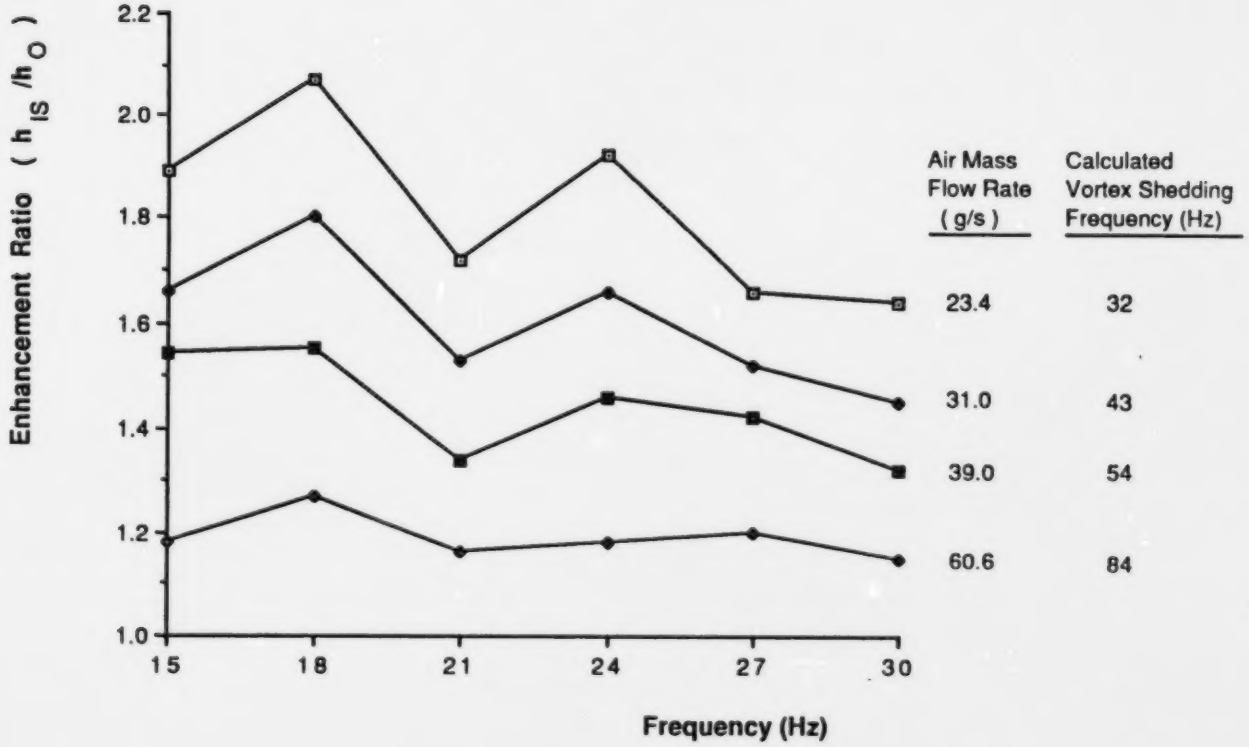


Figure B4 - Influence of Infrasound Intensity on Convection Heat Transfer

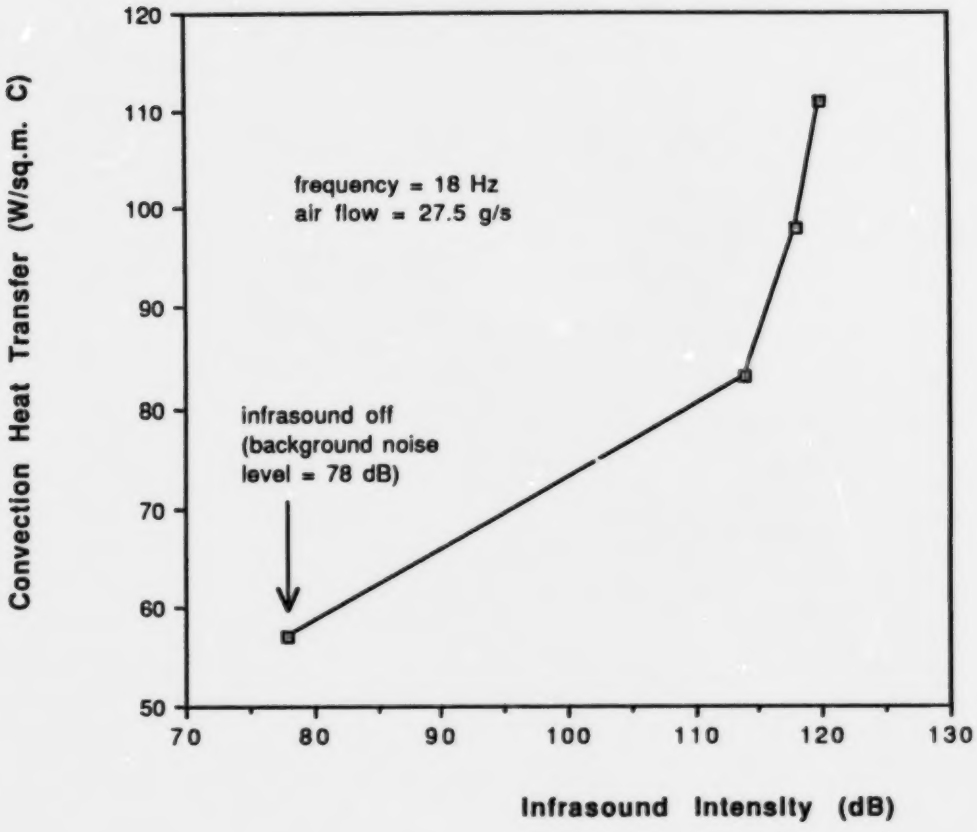


Figure B5 - Theoretical and Measured Forced Convection in Steady Flow

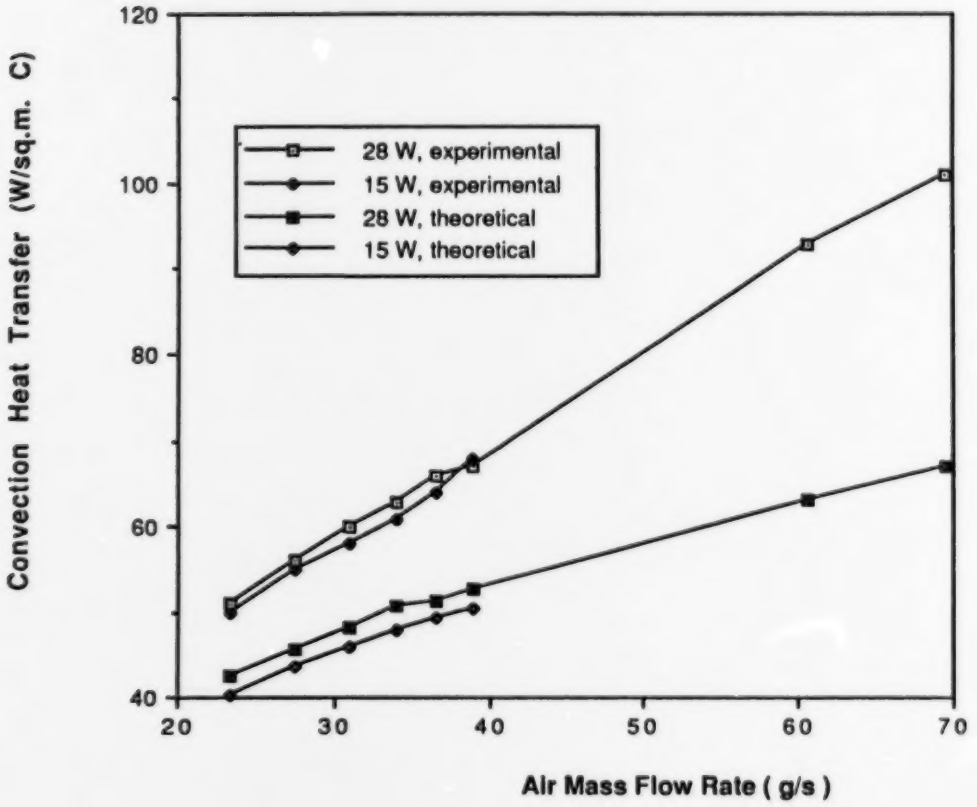


Figure B6 - Infrasonic Enhancement of Convection Heat Transfer (Cross Flow Closed)

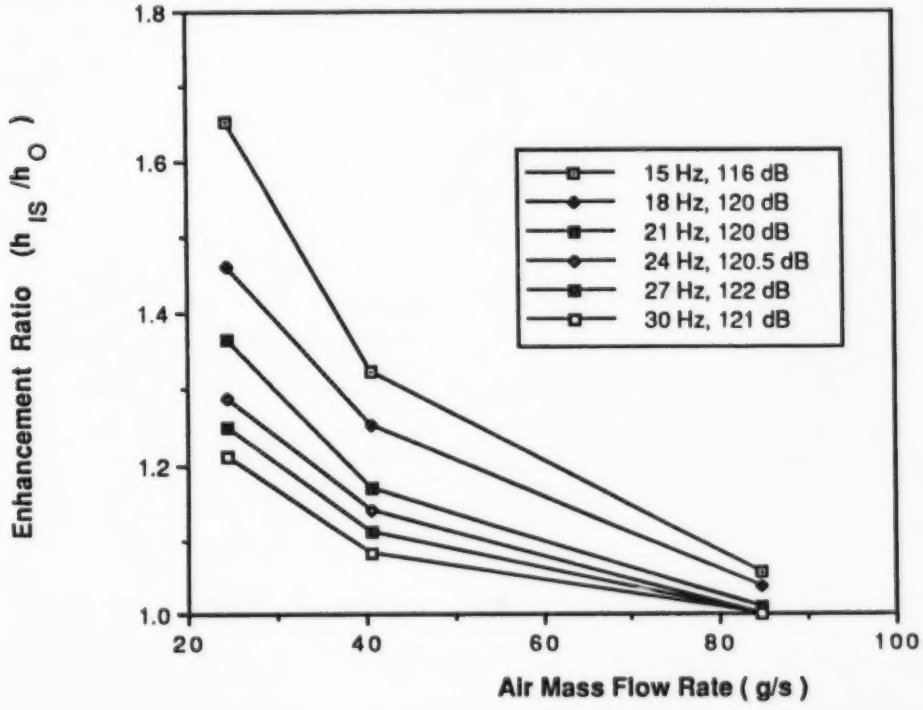


Figure B7 - Comparison of Heat Transfer with Cross Flow Closed and Cross Flow Open Configurations